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JOURNAL BEARING EXPERIMENTAL EVALUATIONS AND DATA CORRELATION

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ABSTRACT

Design of journal bearings for HVAC compressor applications ultimately relies on the collection and analysis of actual test and performance data to confirm design and material selection based on computer modeling and other design methods. A bearing test system which is capable of evaluating full-size journal bearings was utilized in a verification study for a finite element based bearing design code by experimentally determining the oil film pressure distribution in the region of highest expected pressure. A description of the test rig design is given. Experimental results obtained with the bearing aligned and then intentionally misaligned are presented and compared with the pressure distributions predicted by the finite element bearing design code. The code was found to reliably predict the bearing pressures when critical test configuration parameters were supplied.

INTRODUCTION

Industry experience indicates that there is poor correlation between results of bulk material friction and wear tests and the field performance of journal bearings fabricated from the tested materials when installed in refrigerant compressors (Ref.). More advanced materials screening methods are required to ensure that recommended materials provide adequate performance in the intended application. Although the use of instrumented test compressors is possible, compressor testing is costly, time consuming and often does not yield the desired detailed bearing performance data. A need was thus identified for a test method/facility which would closely simulate the application environment and provide real time data on journal bearing performance without the high cost of instrumented compressor tests.

In designing the described test rig the following system requirements were considered:

- ① The test rig should test actual components as opposed to bulk materials.
- ② The test system should be able to approximate the compressor environment including temperatures, pressures, lubrication method, bearing clearance, etc.
- ③ Parameters including shaft speed, shaft alignment and applied load must be controlled and monitored.
- ④ The test system should be flexible and "generic" so that the results obtained will apply to journal bearings installed in a range of products.
- ⑤ The test system should provide access for instrumentation as required for specialized testing and for design code verification studies.

TEST FACILITY DESCRIPTION

The test facility utilized for journal bearing performance studies is pictured in Figure 1 with a diagram of the test chamber shown in Figure 2. The journal bearing is machined to include an oil distribution slot, and is pressed into a support pedestal containing internal passages for the delivery of lubricant from a low pressure oil delivery system to the bearing surface. The pedestal also contains an internal passage for a thermocouple to record changes in the bearing temperature. Modified pedestals can be fabricated for various size bearings or for experiments requiring additional instrumentation such as the oil film pressure measurements described below. The pedestal is attached to the top of a sealed test chamber secured to the crosshead of a servohydraulic universal test machine. A test shaft section is clamped between the inner races of two spherical roller bearings supported by a yoke attached to the test machine load ram. As seen in Figure 2, the test shaft is aligned radially by mating shoulders on the ends of the test shaft, and on the roller bearing inserts. It was found that the accurate machining of these mating surfaces, and their concentricity with the outer diameters of the rotating components was critical to the satisfactory performance of the test system since any eccentricity resulted in an unwanted dynamic load component as the applied load was increased. The rig design allows the hydraulic test machine ram force to be transmitted through the yoke and load the test shaft against the journal bearing following a programmed load profile. The applied load is sensed by a loadcell bolted between the test machine load ram and a ram extension which is threaded into the yoke. The shaft is rotated by a drive motor acting through a drive quill extending from one of the roller bearings to a flexible coupling and continuing through the chamber wall.

Like the support pedestal, the yoke which supports and aligns the shaft can also be modified to meet specific test requirements. Because of the importance of system and bearing alignment in the oil film pressure study described below, five eddy current proximator probes were incorporated into the yoke as shown in Figure 2 to monitor the relative positions of the yoke and pedestal. The outputs of these sensors are used to aide in determining initial system alignment as well as monitoring changes in alignment and relative bearing/shaft movements as the test bearing is loaded.

The test facility also includes a digital data acquisition system and computer for real time display and plotting of selected test parameters and instrumentation outputs, as well as data storage and retrieval. The associated custom software can be modified to accommodate specific test and instrumentation requirements. All acquired data are stored in ASCII files which can later be transferred to a PC for further analysis.

EXPERIMENTAL PROGRAM

Having a test rig in which the configuration and loading can be accurately controlled and monitored presents an opportunity to assess the adequacy and accuracy of bearing performance prediction codes. One such code, the Integrated Bearing Design and Analysis (IBDA) code was an excellent candidate for such a study. Oil film pressure distribution and minimum film thickness variation with applied loading and degree of misalignment are parameters which are essential for assessing a bearing design. The rig was therefore instrumented to measure oil film pressures at nine locations in the high pressure region of a journal bearing for aligned and misaligned configurations. Pressure measurements were made by incorporating miniature pressure transducers into the bearing support pedestal with 0.030 in. (0.76 mm) diameter oil access holes drilled through the bearing wall. Three transducers were placed 20° apart along the bearing midline as well as three 20° apart 0.20 in. from both the front and rear edges of the bearing. Typical transducer installations at the bearing midline are shown in Figure 2. The test program was performed using standard journal bearing materials, hardened steel shafts, a nominal radial bearing clearance of 0.0005 in. (0.013 mm), and mineral based refrigeration oil.

Test System Alignment

The control and measurement of shaft/bearing alignment was of major importance during this test program since the effect of bearing alignment on the oil film pressure distribution was to be investigated. Shaft alignment was controlled by inserting shims under one end of the bearing support pedestal shown in Figure 2 to tilt the bearing relative to the shaft axis of rotation. Although the test rig incorporates a stiff load-train, since the system is still somewhat self-aligning, the actual misalignment under load was anticipated to be less then the static misalignment. This was confirmed by utilizing the three axial proximator probes shown in Figure 2 to record the front-to-rear movement of the yoke relative to the pedestal as the bearing was loaded. This movement increased as the static misalignment was increased, with the direction of movement consistent with movement toward self-alignment.

Test Description

A series of tests were performed to measure oil film pressure distribution as a function of applied load and as a function of shaft alignment. Neither the test chamber nor support pedestal were externally heated so the bearing temperature was allowed to rise due to frictional heating as the applied load increased. The system was run for 30 minutes with a slight preload to allow the bearing to reach a stable temperature prior to the start of the test. The applied load was then incrementally increased to the final load of 700 Lbs (3,100 newtons), with 5 minute soaks at each load to allow the bearing temperature and the oil pressures to stabilize. The static misalignment with no applied load was varied from 0 to +0.4 milliradians (mrads) with positive misalignment acting to increase the bearing clearance at the rear edge of the bearing.

Test Results

The data from the three axial proximitors were used to compute any changes in the preset static axial misalignment as the load was applied. The change in alignment due to the application of a 700 Lb. load was found to be -0.04 mrads for the aligned bearing and -0.25 mrads for the misaligned bearing, i.e. the initial 0.4 mrad misalignment was reduced to 0.15 mrad.

Figure 3 shows typical output data plotted against test run time for the aligned and misaligned tests. For clarity, only pressures at three locations are shown. As can be seen from the plot for the aligned case (Fig. 3a) the pressures at the front and rear of the bearing are approximately equal at the 90° (top dead center) radial location for any given applied load, whereas for the misaligned case (Fig. 3b) the pressure at the 90° radial location is higher at the front than at the rear for all loads. A complete data set also included the output from the remaining pressure transducers and from the proximitors as well as run

conditions such as temperature and shaft velocity. Table 1 includes the complete set of oil film pressures measured at an applied load of 700 Lbs. for both an aligned bearing and a bearing with 0.15 mrad of measured misalignment when under load. Table 1 also includes pressures predicted by the IBDA code which will be discussed later.

Both the test journal bearing and shaft were inspected part way through and at the conclusion of the test program to determine the actual bearing clearance and to check for bearing wear. The inspection results revealed that while the shaft diameter was uniform, the bearing diameter was larger near the ends of the bearing than at the centerline and that the diameter increased slightly during the test program. The first inspection (msmt1) showed that the specified radial clearance of 0.0005 in. was actually 0.00042 in. at the bearing midline and 0.00052 in. near both the front and rear edges. The final inspection (msmt2) showed that the midline clearance had increased by 50 μ in. while the front and rear clearances had increased by 30 and 80 μ in. respectively. The variations in diameter are assumed to be the result of wear caused by startup, misalignment of the shaft during the test program, and misalignment during set-up runs.

IBDA CODE VERIFICATION STUDY

The IBDA code that was used for predicting the dynamic performance of the bearing is a finite element based lubrication analysis which has been developed for application to HVAC products. It is capable of treating variable installation and operational bearing/journal misalignment, and elastic and global bearing support structural deflection. Pre- and post-processors are incorporated to facilitate input and output of data. Parameters predicted by the IBDA code include oil film pressure distribution, oil film thickness, temperature, density, flow rate, and power consumption. Of major concern in this study is the oil film pressure distribution at the nine measurement locations for the aligned and misaligned bearing.

The verification plan was to run the code for the nominal case with no misalignment, compare the predicted pressures for an applied load of 700 Lbs. with aligned bearing test data and identify critical input parameters. The identified critical parameters were then adjusted and the code re-run to assess the effect of the input parameter variation on the correlation with the experimentally measured pressure distribution. Adjustments were to be made based on measurements of the rig geometry, stiffness and alignment. When measurements were not available, e.g. the horizontal load, the sensitivity of the results to these inputs would be examined. Once the ability of the code to predict the pressure distribution for an aligned bearing had been determined, varying degrees of bearing misalignment would be input for comparison to the test data for a misaligned bearing.

Input Parameters

The input parameters required for the code, assuming a rigid structure, are those defining the geometry of the bearing (including the gap and alignment), the applied forces, the rotational speed, oil properties, bearing boundary conditions, and the computational mesh parameters.

With all the rig details including transducer locations known, a non-uniform grid with 15 lengthwise and 36 circumferential mesh points and 14 lengthwise elements was designed and generated by a pre-processing program. In this grid the four outer elements on each side of the bearing have half the lengthwise spacing of the center elements to better accommodate the large pressure gradients which can occur at the outside edges of a misaligned bearing. The computational step was set at 1°. The grid was referenced to the bearing 0° position with the oil groove located at 180°, (90° beyond TDC) and modeled by setting the conditions along the 180° gridline to the 2 psig (13.8 KPa) supply pressure and 75°F (24°C) supply temperature. The boundary conditions for the oil were selected as 0 psig and 75°F along the bearing edges. The viscosity of the oil was determined from the calculated local oil temperature using an analytical formula within the code.

The assumptions that the bearing rig was rigid and provided a controlled test environment for the bearing led to the initial analyses being made with nominal input values. However, checkout tests on the rig indicated that the rig was imposing other constraints which significantly affected the results. These rig constraints can be considered as pertaining to the interdependence of geometry, misalignment and compliance. For instance, clearance variation across the bearing length can be considered to contain a misalignment component and is also affected by compliance of the rig structure.

As stated above, two measurements of the bearing inner diameter and shaft outer diameter were performed during the test series and are identified as (msmt1) and (msmt2). In each of the measurements the radial bearing clearance was evaluated at three stations along the length and second order polynomials were curve-fitted to give clearance values at intermediate stations across the bearing length. The initial code input parameters included the uniform 0.0005 in. nominal radial

clearance. For subsequent cases the calculated bearing clearances were used to determine the clearance at each mesh point and were provided as input to the code.

In a perfectly aligned static structure the applied vertical force is balanced by a vertical reaction force with no horizontal components. However, in a perfectly aligned rotating system the point of maximum film pressure (reaction force) is a few degrees beyond TDC. If the system remains rigid and perfectly aligned as in the case of the test rig, then there must be a horizontal force component acting normal to the axis of rotation whose magnitude is not immediately obvious and which is difficult to evaluate. Also, if there is even a small amount of misalignment in the loading line-of-action then the resultant force can be in any direction in the horizontal plane. The affect of a horizontal force on the predicted film pressure distribution was studied by including a horizontal force acting normal to the axis of rotation in the code input parameters. All other horizontal forces which might be imposed on the bearing by angular or offset misalignment of the shaft and bearing were minimized by careful set-up and the use of the proximitor probes described above and were not included in the input parameters.

The ranges of input parametric values selected during this study resulted in more than 12 cases being run. The inputs for the 12 more significant cases are shown in the upper portion of Table 1.

Oil Film Pressure Prediction Evaluation

The predicted pressures for sample cases are given in Table 1. These show the progression of analyses to arrive at the best fit to the two sets of test data. The percentage differences between the predicted and measured pressures at each location are given in the lower section of Table 1. In order to assess the overall agreement between the predicted and measured pressures, the weighted average difference was calculated for each case using the following formula.

$$\bar{W} = \frac{\sum_{i=1}^9 (P_{mi} - P_{pi}) P_{mi}}{\sum_{i=1}^9 P_{mi}}$$

where:

\bar{W} = weighted average difference

i = measurement location

P_m = measured pressure

P_p = predicted pressure

These values for each case are presented at the bottom of Table 1. The weighting was done to emphasize the variation of the higher pressures, since these are the most important together with the fact that the largest experimental uncertainty occurs at lower pressures. Also the largest analytical uncertainty tends to occur at the bearing boundary because of both low pressure and the difficulty in predicting leakage effects. Obviously, the lower the value the better the pressure distribution prediction.

Aligned Bearing Results

Case 1 (Table 1) assumes a uniform clearance of 0.0005 in., a vertical load of 700 Lb and no horizontal loading. The highest predicted pressure underestimates the measured value by 32%. In addition the predicted maximum pressure is located at about 95°, whereas the measured maximum pressure occurs at about 105°. \bar{W} for this case was 44% reflecting the poor prediction at high pressures.

Case 2 was run to show the effect of using the initial clearance distribution measurement (msmt1) with a midline clearance of 0.00042 in. instead of the nominal 0.0005 in. uniform clearance used in case 1. The predicted maximum pressure has risen 39% as a result of this change, and is only 6% below the measured value although its location has not changed. This improvement in maximum pressure prediction lowers \bar{W} to 28% (see Table 1). Case 3 shows the effect of using the final clearance measurement (msmt2) which slightly decreased the clearance across the entire bearing length compared with msmt1. Similar results were obtained but with more asymmetry in the pressure distribution reflecting the asymmetry in the clearance.

The difference in the measured vs. predicted location of the maximum pressure indicated that there was a negative horizontal load being applied by the rig. Parametric analyses for that load were performed and 100 Lbs was found to shift the pressure distribution to give results closest to the measured values. Cases 4 and 5 were run to show the effect of the 100 Lb horizontal load with clearance measurements msmt1 and msmt2 respectively. The pressure distribution for Case 4 yields the lowest value of \bar{W} with both the position and magnitude of the maximum pressure closely predicted, as well as the other midline pressures. The importance of proper input parameters is seen by comparing this prediction to Case 1 which underestimated the maximum pressure by 32% using the nominal value for clearance and assuming no side loading.

Figure 4a shows a graphical representation of the pressure distribution in the high pressure region with shaft rotation being from left to right. The film pressures predicted by Case 4 are displayed as contours. The locations of the pressure transducers are also displayed along with the associated measured pressures. The contour plot demonstrates the high pressure gradients occurring along both the axial and circumferential axes. The predicted pressure gradient across the 0.030 in. transducer opening at the highest gradient region is approximately 115 psi in the axial direction and 180 psi in the circumferential direction. The low measured values compared to the prediction at the front and rear of the bearing at 110° may be the result of uncertainty in the true transducer location or local clearance variation not picked up by the 'three station' inspection measurement.

Misaligned Bearing Results

An initial calculation using the measured static misalignment value of 0.4 mrad indicated that the misalignment at load was much less than 0.4 mrad. Cases 6 and 7 were run to investigate the effect of a set amount of misalignment (-0.05 mrad) on the pressure distributions of Cases 4 and 5. They show higher pressures at the front of the bearing than at the rear, which is consistent with a negative misalignment angle and the midline pressures have decreased slightly compared to Case 4. The large values of \bar{W} reflect the difference between the predicted pressures with negative misalignment and the measured values with positive misalignment.

Cases 8, 9, and 10 were run with a code input misalignment of 0.075 mrad, with and without the -100 Lbs horizontal load. The values of \bar{W} were significantly reduced but some of the predicted pressures at the front of the bearing where the misalignment increased the clearance exceeded the measured values by a significant amount. Inspection of the distributions indicates a smaller horizontal load and more misalignment may be present. Hence the cases 11 and 12 were run with a horizontal load of -80 Lbs and misalignments of 0.1 and 0.15 mrad. As seen in Table 1 the values of \bar{W} are further reduced to 31% and 29% respectively. Case 12 using the clearance measurement msmt2 gave the closest results with the value of \bar{W} similar to the result for the best aligned case (Case 4). The input misalignment of 0.15 mrad for this case is in agreement with the measured misalignment from proximitors readings.

Figure 4b shows a contour plot for the best misaligned bearing fit (case 12) similar to that shown for the best aligned case. The combination of a higher predicted maximum pressure compared to the aligned case and the shift in maximum pressure location has resulted in very high pressure gradients in the lower right quadrant of Fig.4b.

CONCLUSIONS

1. The experimental results obtained during the test program confirm the IBDA code is capable of accurately predicting both the magnitude and location of the maximum oil film pressure for both aligned and misaligned journal bearings if the proper code input parameters are known and applied. For the best cases the maximum oil film pressure is predicted to within 3% of the measured value for an aligned bearing, and within 10% of the measured value for a misaligned bearing.
2. The predictions of pressure distributions for both the aligned and misaligned bearing are somewhat less accurate than the predictions of the maximum pressures. The pressures at all measurement locations are conservatively predicted for both the aligned and misaligned cases. The oil film pressure gradients are extremely high, on the order of 5000 psi/in and small variations in the pressure measurement locations relative to the assumed analytical grid location can lead to large differences between measured and predicted pressures.
3. The code input misalignment of 0.15 mrad for the best misaligned prediction is in agreement with the experimentally measured misalignment.
4. The critical code input parameters identified include all load vectors, bearing clearance and clearance distribution, and shaft misalignment.
5. The capability of the described test rig to establish, control, and record parameters such as misalignment, load, and oil film pressure make it a viable system for future bearing performance related studies.

REFERENCE

Cusano, C. et al, Accelerated Screening Methods for Predicting Lubricant Performance in Refrigeration Compressors, Final Report, DOE/CE/23810-45, 1994

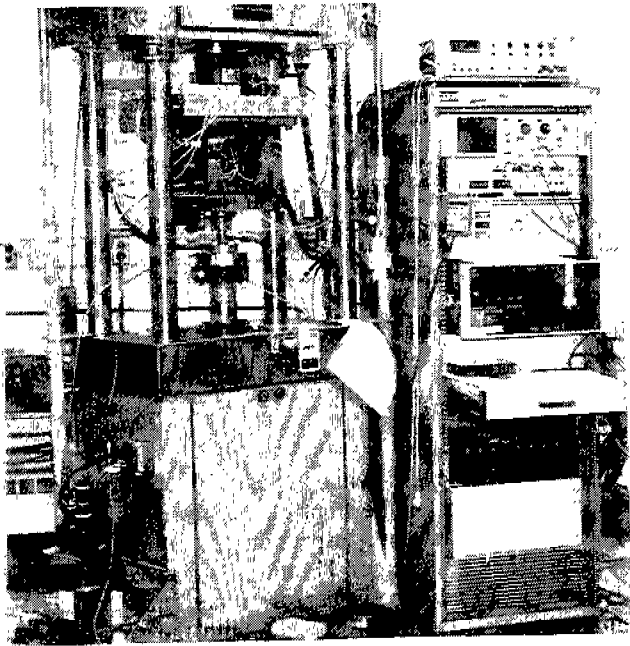


Figure 1 Journal Bearing Test Facility

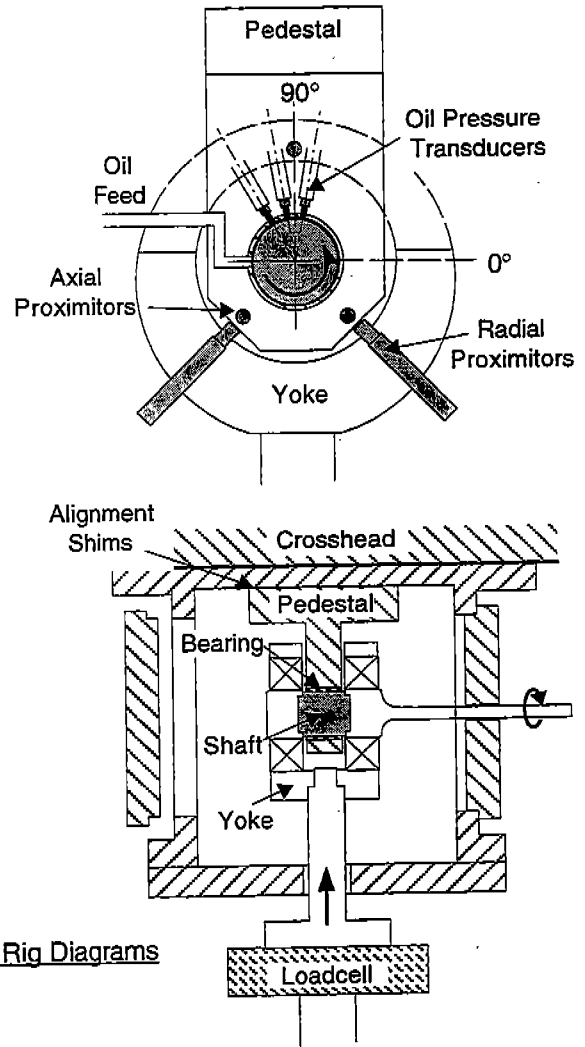


Figure 2 Test Rig Diagrams

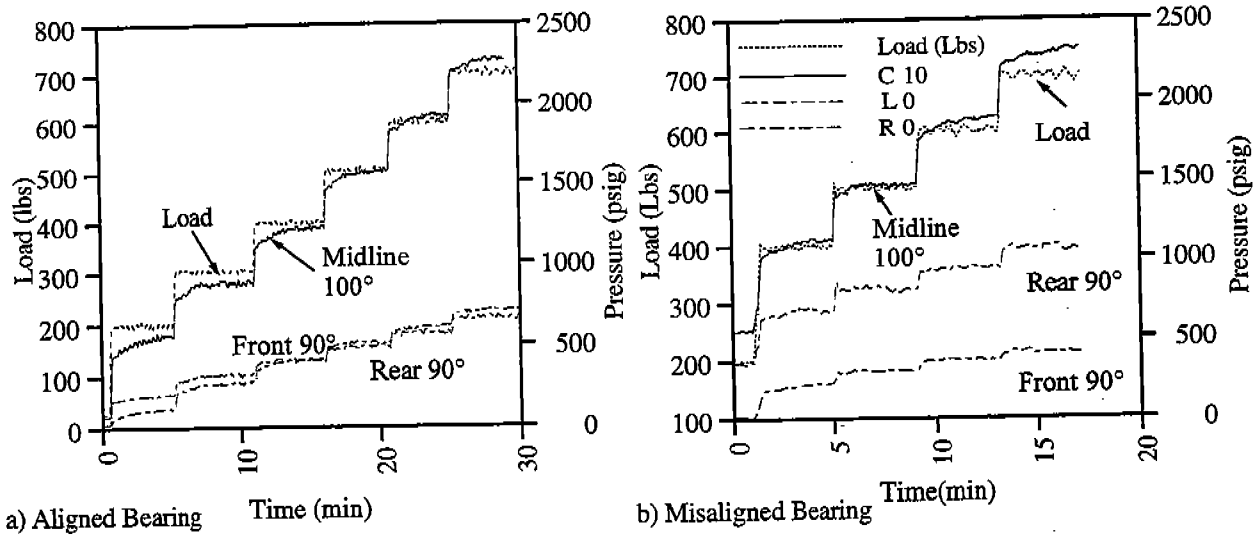
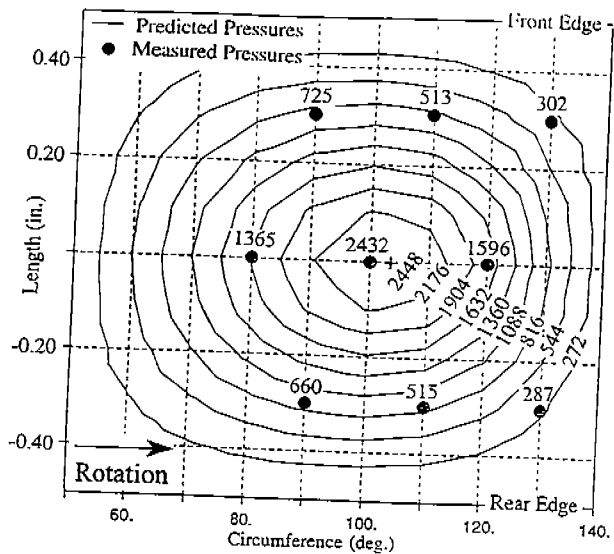


Figure 3 Typical Test Data

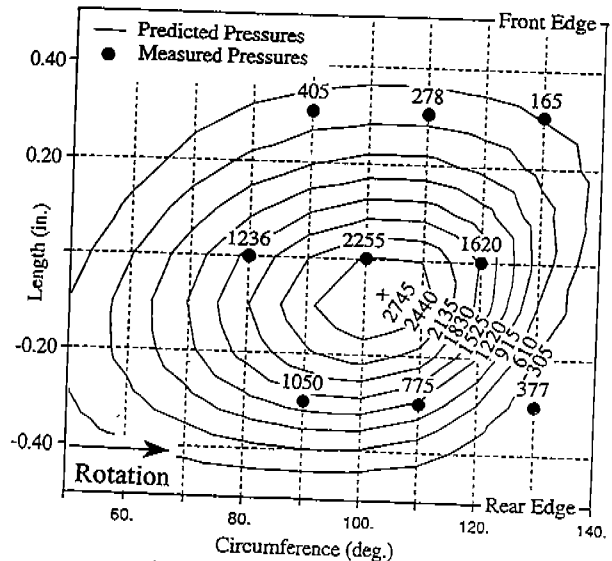
Table 1 IBDA Code Input Parameters and Output Pressures at Sensor Locations and Comparison with Test Results

CODE INPUT VALUES														
CASE #	ALIGNED						MISALIGNED							
	1	2	3	4	5		6	7	8	9a	9b	10	11	
PARAMETER														
Horizontal Force Lbs (Fx)	0	0	0	-100	-100		-100	-100	-100	0	-100	-80	-80	
Vertical Force Lbs (Fy)	700	700	700	700	700		700	700	700	700	700	700	700	
Misalignment θ_x millirads	0	0	0	0	0		-0.05	-0.05	0.075	0.075	0.075	0.1	0.15	
Misalignment θ_y millirads	0	0	0	0	0		0	0	0	0	0	0	0	
Clearance mil	0.5	msmt1	msmt2	msmt1	msmt2		msmt1	msmt2	msmt1	msmt2	msmt2	msmt2	msmt2	
PRESSURES (psig)														
LOCATION	MEASURED	PREDICTED				MEASURED	PREDICTED							
Front 90°	725	1095	918	1003	903	972	405	1200	1247	593	708	675	605	463
Front 110°	513	924	713	796	891	1011	278	1131	1264	647	600	737	643	522
Front 130°	302	270	146	117	318	323	165	283	293	339	160	340	304	301
Mid 80°	1365	1440	1905	1960	1658	1651	1236	1603	1578	1569	1972	1656	1697	1610
Mid 100°	2432	1604	2237	2488	2448	2659	2255	2386	2517	2348	2525	2681	2612	2482
Mid 120°	1596	947	1161	1243	1780	2028	1620	1761	1919	1742	1270	2050	1864	1804
Rear 90°	660	1093	911	714	901	704	1050	655	538	1311	1003	986	1112	1382
Rear 110°	515	915	702	535	882	680	775	699	543	1209	702	909	960	1167
Rear 130°	287	261	134	83	306	217	377	330	229	254	21	108	127	69
PERCENTAGE DIFFERENCE BETWEEN PREDICTED AND MEASURED PRESSURES														
Front 90°		51	27	38	25	34		196	208	46	75	67	49	14
Front 110°		80	39	55	74	97		307	355	133	116	165	131	88
Front 130°		-11	-52	-61	5	7		72	78	105	3	106	84	82
Mid 80°		5	40	44	21	21		30	28	27	60	34	37	30
Mid 100°		-32	-6	5	3	12		6	12	4	12	19	16	10
Mid 120°		-41	-27	-22	12	27		9	18	8	-22	27	15	11
Rear 90°		66	38	8	37	7		-38	-49	25	-4	-6	6	32
Rear 110°		78	36	4	71	32		-10	-30	56	-9	17	24	51
Rear 130°		-9	-53	-71	7	-24		-12	-39	-33	-94	-71	-66	-82
WEIGHTED AVE.		44	28	26	25	28		71	88	32	35	39	31	29
KEY: Value within ± 15% of measured														

KEY: Value within +/- 15% of test result



a) Aligned Bearing (Case 4)



b) Misaligned Bearing (Case 12)

Figure 4 Comparison of Measured and Predicted Pressures for Aligned and Misaligned Bearing